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HYDRODYNAMIC OPTIMIZATION OF THE TRIBOLOGICAL SYSTEM PISTON - RING - LINER IN RECIPROCATING COMPRESSORS

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ABSTRACT

Mechanical friction losses in reciprocating machines amount to approximately 60% in the tribological system 'piston-ring-liner'. Improvements by proper design of the ring profile can contribute to optimize this tribological system.

To improve hydrodynamic lubrication conditions in the piston-ring-liner assembly with regard to frictional losses and prevention of wear, sliding profiles must be optimized.

Theoretical and experimental results concerning various ring profiles are compared for steady state and nonsteady state conditions. In both cases good correlation between theoretical and experimental results can be stated. In order to optimize piston ring profiles with regard to smallest friction losses and highest load capacity the application of the developed simulation programs can be useful. By this means a prediction of reduced energy losses and prevention of wear can be made.

INTRODUCTION

Due to the rising prices of the primary energy sources and their limited reserves, an increase in the efficiency of reciprocating machines has become more and more important. Over 60% [1] of the mechanical engine losses arise in the construction group piston-piston ring-liner in reciprocating machines. The piston rings contribute substantially to these losses. The optimization of the piston ring profile regarding an increase in the hydrodynamic lubrication area as well as the decrease in the friction results is unavoidable, if an improvement in the operating conditions should be achieved.

In order to make a statement about the hydrodynamic behavior of piston rings, a clarification of the lubrication conditions of this tribological system is necessary. This applies especially to a piston fitted with several piston rings, where the reaction of the rings with each other during operation must be investigated.

CALCULATING MODELS

Only a few calculating models exist to determine the lubrication conditions of the piston-ring-liner system in refrigerant compressors. During the

Purdue Compressor Conferences 1974, 1976 and 1978 Kruse [1], Wrede and Kruse [2] and Wrede [3] read the mathematical papers about the solution of this problem. Todsen and Kruse [4] used these models for several piston ring profiles to investigate their influence on the hydrodynamic behavior of piston rings.

The calculating models developed in this paper to describe the tribological system piston-ring-liner, as well as those aforesaid, are based on Reynolds differential equation for incompressible laminar flow.

$$\frac{\partial}{\partial x} \left(h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(h^3 \frac{\partial p}{\partial z} \right) = 6\eta(U_1 - U_2) \frac{\partial h}{\partial x} + 6\eta h \frac{\partial}{\partial x} (U_1 + U_2) + 12\eta v \quad (1)$$

These models suffice to describe the tribological system piston-ring-liner with one piston ring. A statement about the oil supply of the following rings has to be made when using several rings.

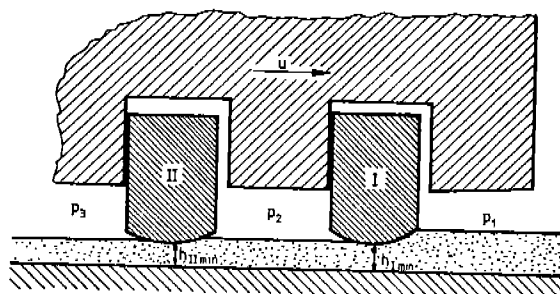


Figure 1. Schematic oil distribution with two piston rings

With regard to this, the boundary conditions for the pressure development at the rising and flowing off end of the lubrication gap must be formulated. The beginning of the pressure forming lubrication film area lies at the dam face of the oil with the beginning of an oil supply which fills the gap.

Here applies:

$$p_y = p_{Gas_0} \quad (2)$$

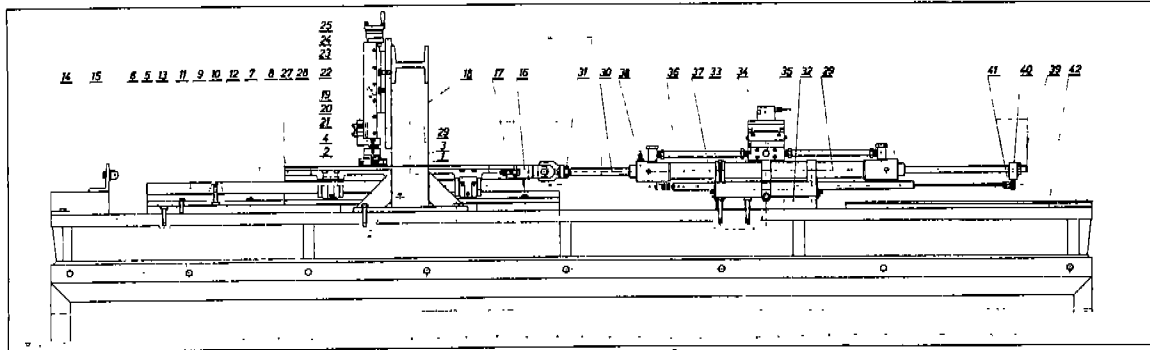


Figure 2. Experimental slider

$$\frac{\partial p}{\partial x} \neq 0 \quad (3)$$

If a tearing of the lubrication film occurs due to a geometrical gap widening at the outlet end of the system, Guembel's boundary conditions can be used.

$$p_y = p_{Gas_u} \quad (4)$$

$$\frac{\partial p}{\partial x} = 0 \quad (5)$$

With the knowledge of the tear position, the oil film thickness at this position can be determined in consideration of the piston ring contour. Through this oil supply for the second ring is also known. Now the calculation of the pressure profile below this piston ring under the given boundary conditions must be carried out so long until a radial balance is given. From this the limitations of the pressure forming lubrication film area of the second piston ring for this crank angle position can be determined. The equivalent applies to the third up to the n-th ring.

EXPERIMENT EQUIPMENT

The experimental tests were carried out on a model slider test rig as well as in a test engine. The test rig has already been introduced by Wrede /2/ 1976 at the Purdue Compressor Conference. The model slider has been designed as even slider. The even piston is hung up dynamically in a portal and can take up to three piston strips. The sliding plane can be moved at a constant speed as well as with a speed expulsion corresponding to the crank gear kinematic.

To measure the lubrication conditions at the piston, piston-ring-liner assembly tests were carried out using various shaped piston ring models. Since representative signals were required in order to compare theoretical and experimental results various ring models were used.

The ring profiles are shown in Fig. 3

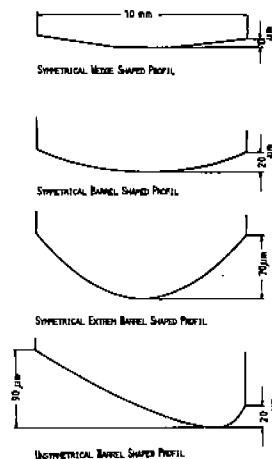


Figure 3. Sliding profiles

First the wedge shaped profile is shown (angle of wedge = 0,1° deg, height of wedge = 10 · 10⁻⁶ m). The barrel-shaped profile is symmetrical, the height of profile is 20 · 10⁻⁶ m in this case. Thinking of the extreme barrel-shaped ring this height amounts to 70 · 10⁻⁶ m. The last profile shown in Fig.3, is the unsymmetrical barrel-shaped one. The height at one side of the ring will be 90 · 10⁻⁶ m and 20 · 10⁻⁶ m at the other side.

To check the effect of a gas pressure on the piston rings on the lubrication conditions, tests had to be carried out on a complete machine. These tests were carried out on a crosshead-type engine. During the running of this machine with starting air gas pressure stresses of the piston ring occurred, which are similar to those of refrigerant compressors.

The knowledge of the minimum oil film thickness and of the lubrication film pressure is essential for the judgement of the lubrication film conditions. To obtain a dependable oil film thickness measurement, inductive gauges were applied in the cylinder box of the crosshead-type engine as on the model slider,

- with the disadvantage of not being able to measure continually over the stroke,
- with the advantage of a greater reliability compared to other measuring techniques.

There are difficulties with the distance measurement of having to form the difference of two almost equally high values, besides this, a tipping of the piston ring cannot be recorded. Capacitive transmitters - steamed up or inlaid - produce difficulties through the fluctuations of the dielectricum and offer no protection against abrasion.

The development of the cylinder liner in Figure 4 shows the arrangement of the measuring positions. The six measuring bores, 3 on the clutch and contro clutch sides, respectively, in which the pressure and distance gauges are installed, are divided into 3 measuring planes:

- Measuring level 1 is found near the top dead centre of the piston. It is positioned, so that even the lowest ring coats the measuring transmitters when all rings are applied.
- Measuring level 2 lies in such a place, so that the pic-ups still lie above the 6 lubrication oil taps.
- Measuring level 3 is found in the area of maximum piston speed above the gaps.

The position of the measuring bores in a vertical direction is chosen in such a way, that they lie near the lubrication oil taps and so that the probability of the presence of oil is large also during irregular oil distribution over the circumference.

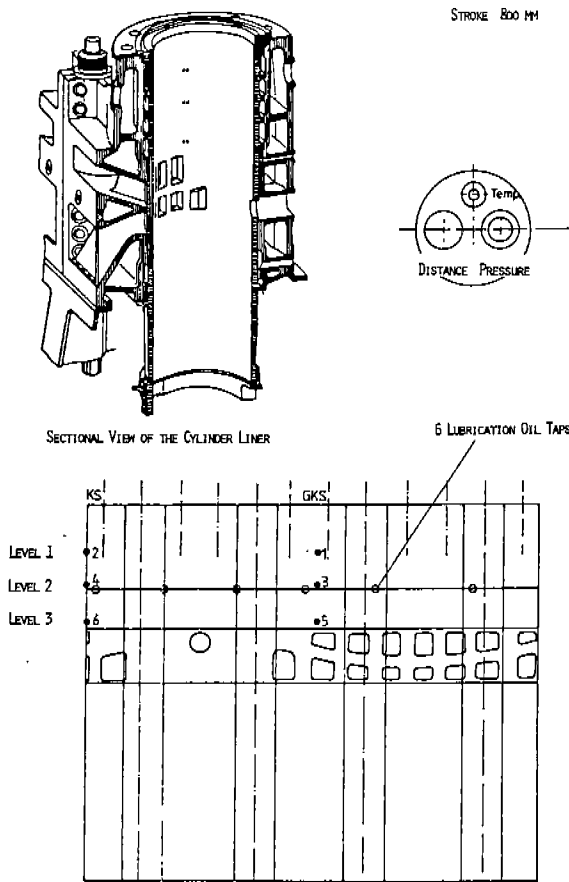


Figure 4. Arrangement of the measuring positions

In Figure 5 the minimum lubrication film level and the maximum lubrication film pressure are shown over the relative speed for three tests with different oil supply. The subscripts of h (lubrication film level) and p (lubrication film pressure) mark the piston rings 1-3. The different speed areas for the piston rings are produced through the piston, which slides past the stationary measuring gauges at an inconsistent speed.

Through this figure the behavior of the piston rings in a ring packet is made clear. While by a normal lubrication (test 1), that is exclusively with the lubricators, the three shapes of the lubrication film lie almost on one curve, which suggests a good lubricant supply of all rings by use of three similar rings, at a limited oil supply (test 3) from the dispersers, almost similar lubrication film levels for the rings are measured, which thereby lie on different curves. A large amount of lubricant (test 2) makes the curve of the previous ring 3 steeper.

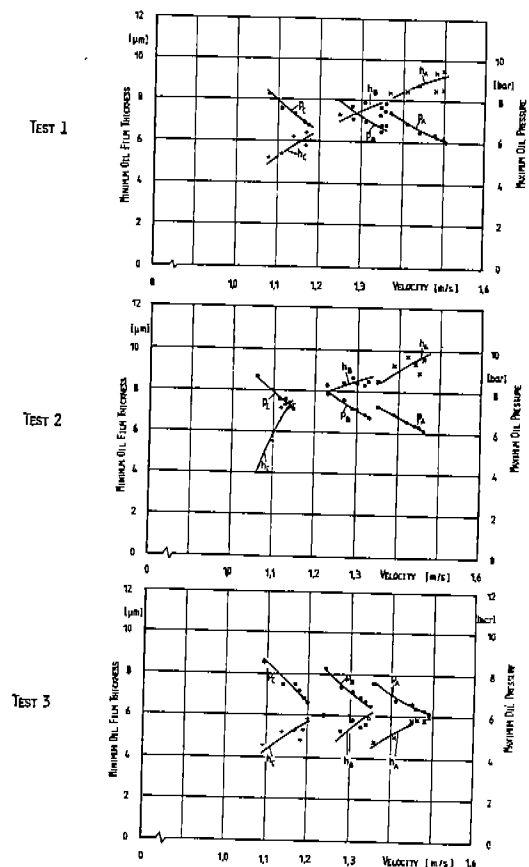


Figure 5. Lubrication film level and pressure versus velocity

From this one can conclude:

- on the one hand, that the lubrication film level

of the previous ring 3 is influenced during increased oil supply (test 2). The curve of this ring becomes appreciably steeper. Both of the following rings also show a slight rise in the lubrication film level.

- on the other hand, that by a sinking oil supply the lubrication film levels of the following rings are strongly influenced. The previous ring is only slightly influenced.

COMPARISON OF THE CALCULATED AND TEST RESULTS

A comparison of the lubrication film level between the values gained on the model slider and the calculated values is shown in Figure 6.

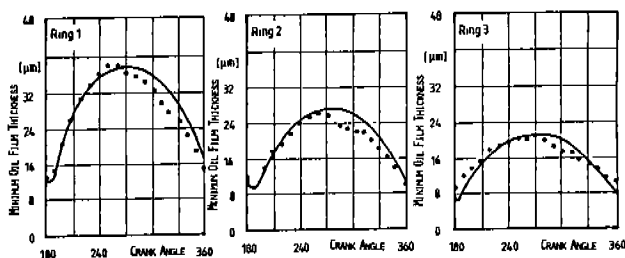


Figure 6. Calculated and measured lubrication film levels in a ring packet (wedge-shaped; barrel-shaped; barrel-shaped) dependent on the crank angle

Here a good conformity is produced in the first half of the stroke, while somewhat lower levels were measured in the retardation phase.

The comparison between the level measured in the test engine by starting air conditions and those levels gained by the calculation programs, should be carried out at two single rings of varied globularity.

The calculated shaped lubrication film level under a piston ring for one revolution is in each case filled in on the diagrams.

Figure 7 shows the theoretical values through a curve and the experimentally gained values with symbols.

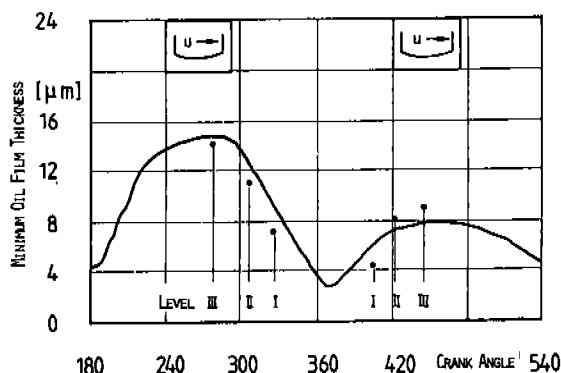


Figure 7. Measured and calculated lubrication film levels on a piston ring in starting air conditions

As the measurement resulted at three fixed places in the cylinder bush liner, only individual values can be used for comparison. A mean is taken here of the inserted data from several readings as also in the following comparisons. The values in the upward stroke are somewhat too low; in the downward stroke the data of the middle and lower dispensers are somewhat too high. The geometric data of the running profile was measured with the help of a surface testing apparatus at several positions of the circumference meaned and then put into the calculating program. These differences are probably produced, because the piston ring has not exactly this form over its circumference.

A better conformity results from the use of a new piston ring (Figure 8).

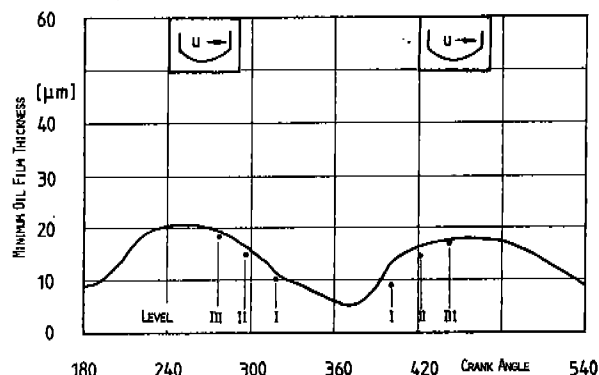


Figure 8. Measured and calculated lubrication film levels on a piston ring in starting air conditions

On comparing the test results

- from the model slider without gas pressure stress of the piston ring and
- from the test engine by various operating conditions

with the theoretical values of the calculating program, the extensive conformity shows the applicability of this calculation model for describing the tribological system piston-piston ring-liner with several piston rings.

USE OF THE CALCULATION PROGRAMS

The influence of the position of the globularity is shown in the following by means of 3 calculations. The ratio of the straight lines L and L_1 was varied, the size of the globularity, the gas pressure stress of the ring here the refrigerant compressors and the level of the piston rings - here 4 mm - remained constant.

Figure 9 shows an eccentric barrel-shaped profile with a length ratio $L_1/L = 0.2$. Through the profile resulting from this, large lubrication film levels are produced in the upward stroke, while only lubrication film levels lying below 10 μm are produced in the downward stroke.

UNSYMMETRICAL BARREL SHAPED PROFIL



$$\frac{L_1}{L} = 0,2$$

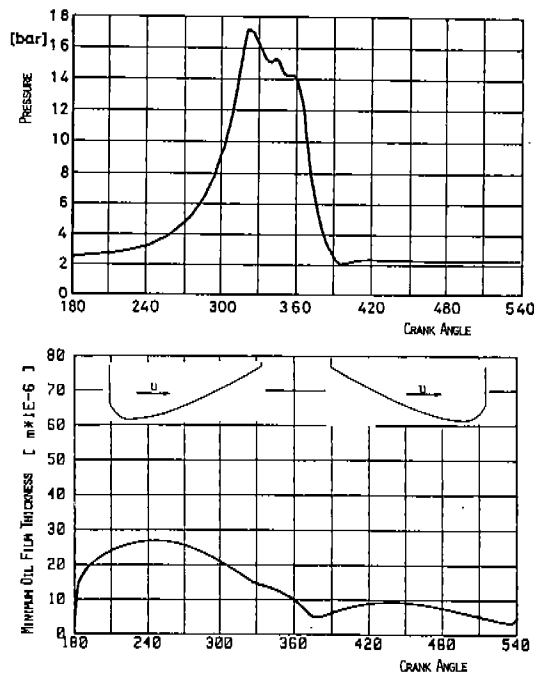


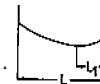
Figure 9. Calculated lubrication film level under an eccentric barrel-shaped piston ring with gas pressure stress of a coolant compressor

Through the displacement of the globularity (Figure 10) the ratio $L_1/L = 0,3$ - the lubrication film level sinks in the upward stroke, but rises in the downward stroke. Despite the gas pressure stress, still higher lubrication film levels are reached here from BDC to TDC caused by the type of profile, than in the opposite direction of operation.

By the displacement of the globularity to the ratio $L_1/L = 0,5$ (figure 11) the influence of the gas pressure stress on a centrally globular profile becomes apparent. The lubrication film levels in the downward stroke now exceed those of the upward stroke.

A comparison of these results clarifies the influence of the position of the globularity: at equal stress the lubrication film level sinks in the upwards stroke, when the globularity is displaced to the middle.

UNSYMMETRICAL BARREL SHAPED PROFIL



$$\frac{L_1}{L} = 0,3$$

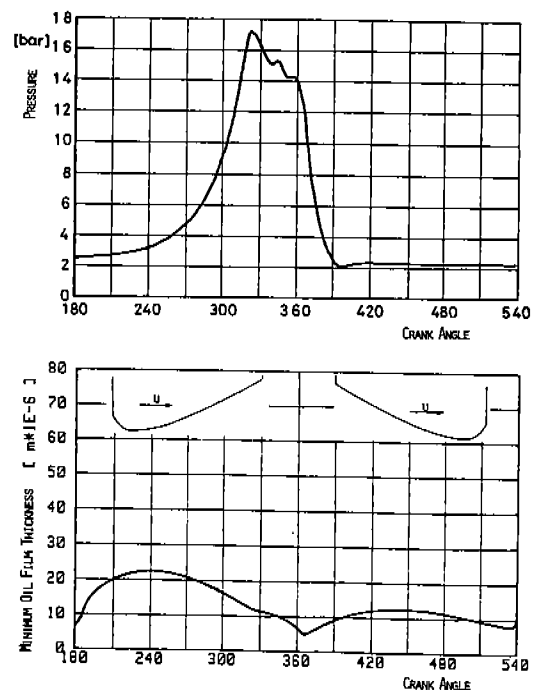


Figure 10. Calculated lubrication film level under an eccentrically barrel-shaped piston ring with gas pressure stress of a coolant compressor

SYMMETRICAL BARREL SHAPED PROFIL



$$\frac{L_1}{L} = 0,5$$

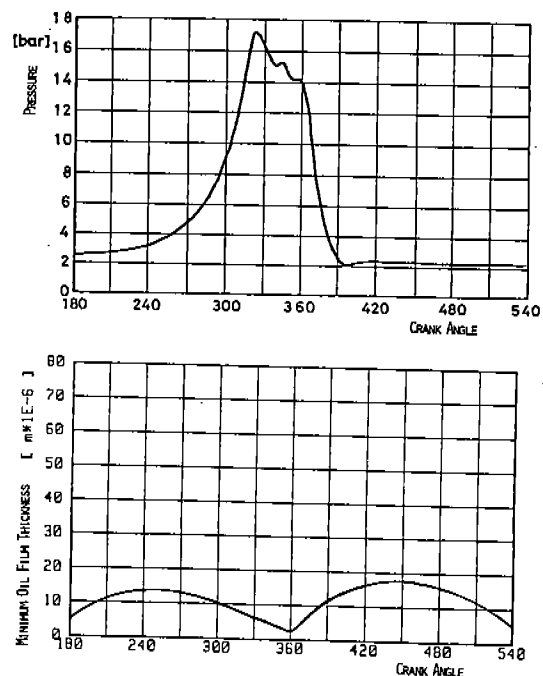


Figure 11. Calculated lubrication film level under a centrally barrel-shaped piston ring with gas pressure stress of a coolant compressor

Due to the preceeding observations an optimization program for a piston ring was worked out. The aim of this program is to determine piston ring profiles with the most regular level possible at the lowest friction output.

The required input data are therefore:

- geometric data of the piston
- piston ring height
- medium piston speed
- dynamic viscosity and density of the oil
- gas pressure

To determine the optimum profile the program will run so long until a profile with the desired characteristics is produced through changing the position and the level of the globularity.

To clarify the speed dependancy of the minimum lubrication film level of the piston rings with only partly filled, convergent gaps a calculation with two equal wedge rings was carried out. The numbers in Figure 12 mark rings 1 and 2. A filling of the first ring of 84% of its gap length is marked with 1', the filling of 55% with 1".

This diagram shows that the increase of the lubrication film level at increasing speed will be less through a part filling of the lubrication gap. Thus, ring 1' no longer reaches the lubrication film level of the following ring with a filling of 84%. With a filling of 53% (2") the second ring shows only a very slight speed dependancy.

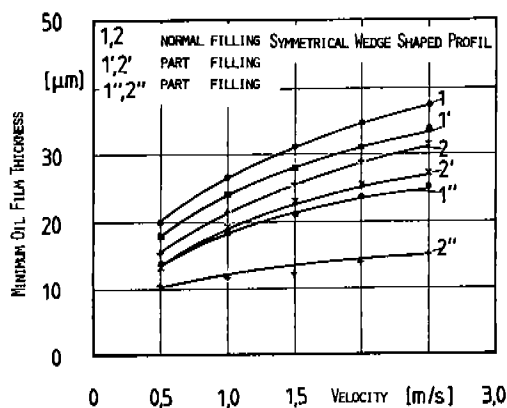


Figure 12. Lubrication film level of two wedge-shaped rings with a varied oil supply

From this it is clear, that the ring with the partly filled lubrication gap not only shows lower lubrication film levels than the completely full one at low relative speeds, but also scarcely rises at increasing speed.

In the following the reaction of the ring packet on the first ring should result through the comparison

of the lubrication film levels of a ring running alone and another in a packet.

The differences which result if you observe the first ring alone or in the packet, are shown in Figure 13. The lubrication film levels were established without gas pressure stress of the piston ring.

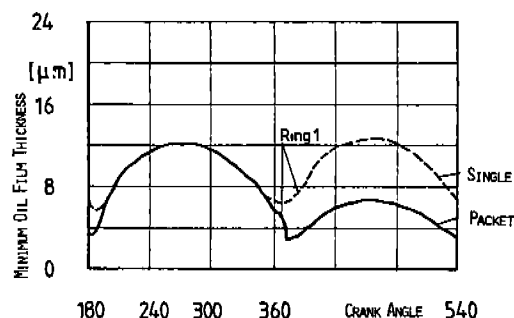


Figure 13. Minimum lubrication film levels under a barrel-shaped piston ring with consideration and neglect of the influence of the ring packet

The influence of the ring packet becomes clear in this diagram during observation of the downward stroke (360 - 540° KW). Very much lower lubrication film levels are reached here, determined by the degressing effect of the previous rings. The decrease is brought about still by the first crank angles of the upward stroke, but which produces no differences for the ring observed alone in the remaining area, as the second and third rings follow.

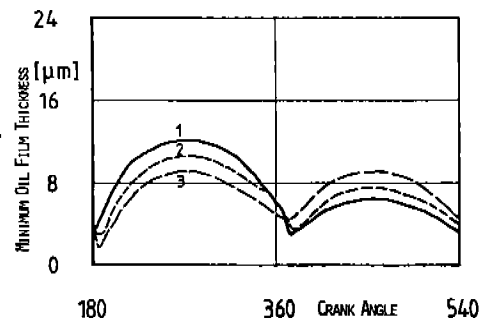
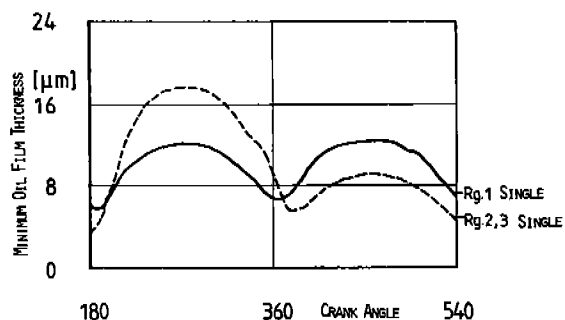


Figure 14. Minimum lubrication film levels in a ring packet with individual lubrication of single consideration of the packet influence

The lubrication film levels of all three rings contained in the packet, on the one hand with individual lubrication, on the other hand with the oil supply in the ring packet are shown in Figure 14. The single rings are shown in the upper half of the diagram, rings 2 and 3 reach greater lubrication film levels as ring 1 through the eccentric running profile in the upward stroke. The lower half of the diagram shows the lubrication film level of these rings considering the packet effect. Here the greater lubrication film level is calculated for the first ring in the upward stroke and for the third ring in the downward stroke.

The influence of an oil stoppage by a large oil supply and an eccentric sliding piston with the result, that oil stops up in the space between piston and cylinder wall, is explained in the following section.

The calculations are carried out for a constant stoppage length in front of the ring, which extends, in the case of the example shown in Figure 15, over a length in the size of the axial piston ring level. Furthermore it is assumed, that the oil pressure increasing through the oil stoppage - as in a step bearing - is also effective behind the piston ring at the same time as stress.

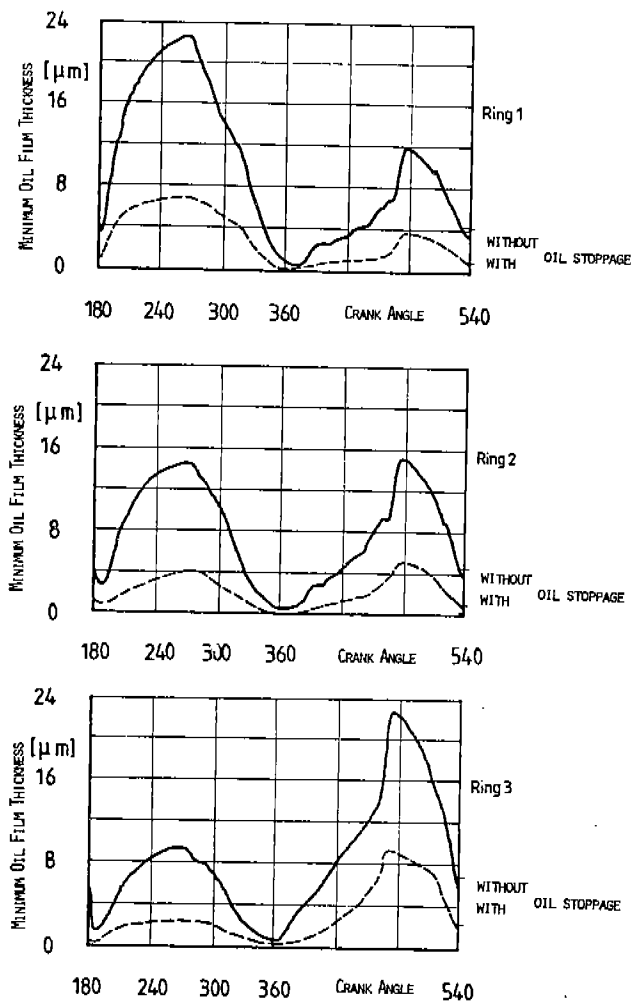


Figure 15. Minimum lubrication film level under the piston rings of a ring packet (with and without oil stoppage)

This extra stress, apart from the gas pressure, causes the lubrication film levels under the first ring to sink considerably, with the result, that the third piston ring only reaches a lubrication film level of $2,3 \mu\text{m}$ at maximum speed. A doubling of the oil stoppage length causes the third ring to fail.

CONCLUSION

In this work calculation models were presented to judge the tribological system piston-ring-liner for a piston fitted with several rings. To check the boundary conditions and to extend these calculation models, test were carried out

- on a model slider test rig and
- on a test machine

to gain the dimensions of lubrication film pressure and lubrication film level, which describe the lubrication film.

The test machine enabled the additional gas pressure admission of the piston rings. These investigations took place on pistons, that were fitted

- with single rings and
- with ring packets.

With this the running profiles of the piston rings, the gas pressure stress, the sliding speeds and the oil supply were varied. Through these tests the behavior of the piston-ring-liner system could be judged under different operating conditions.

The calculation models produce a program system with programs to judge the lubrication characteristics on those pistons, which are fitted with several piston rings

- under nonsteady conditions during reception of very wide rings and
- under steady conditions during consideration of the components of the circumference.

The good conformity of the theoretically and experimentally obtained data confirms the applicability of these calculation models to gain the lubrication characteristics in the piston-ring-liner system. Through this a simple investigation of the influence parameter on the piston-ring-liner system is possible, as is shown by means of a few examples.

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